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FLIGHT PROTOTYPE AMMONIA STORAGE AND FEED SYSTEM

Prepared by

AVCO MISSILES, SPACE AND ELECTRONICS GROUP
SPACE SYSTEMS DIVISION
Lowell Industrial Park
Lowell, Massachusetts 01851

FINAL REPORT

AVSSD-0100-67-RR
Contract NAS5-10128

January 1967

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
TECHNICAL MANAGEMENT
AUXILIARY PROPULSION BRANCH
NASA/GODDARD SPACE FLIGHT CENTER
Greenbelt, Maryland

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APPROVED

T.K. Pugmire

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I. INTRODUCTION

A. PROGRAM OBJECTIVES

The objective of this program was to design, develop, fabricate, and test a flight-prototype ammonia-propellant storage and feed system. The storage system was to provide storage for 57 pounds of ammonia. The storage container was to be adequate for the environmental conditions specified. The feed system was to provide for a maximum flow condition of 248 x 10^{-6} lb/sec for 208 seconds, once per 24-hour period, and a minimum flow rate of 55.5 x 10^{-6} lb/sec for a single pulse of 0.015-second duration. The adjustable regulation system was to provide no more than 1-percent variation of output pressure for the supply pressure associated with the environment temperatures.

B. PROGRAM ORGANIZATION

This program originated from the Auxiliary Propulsion Branch of the NASA Goddard Space Flight Center. Mr. William C. Lund was the Technical Officer for NASA GSFC. The Project Director of Avco SSD was T.K. Pugmire. The other participants in this program were W. Davis, R. Ingemi, and J. Malenda.

C. PROGRAM SUMMARY

A flight-prototype ammonia-propellant storage and feed system suitable for meeting specified environmental qualification was delivered to NASA GSFC. The loaded storage system will meet the ullage and strength requirements for the conditions specified. The feed system provides not greater than 1-percent regulated pressure variation,with either liquid or gaseous ammonia. Principle problems observed in the program were the availability of a satisfactory solenoid valve and the fabrication of a titanium tank.

II. COMPONENT DESIGN AND DEVELOPMENT

A. PROPELLANT STORAGE TANK

Based on the requirement that the storage tank be capable of storing 57 pounds of usable propellant and have adequate ullage volume to satisfy system operating and thermal expansion requirements, the tank was designed to hold 57.5 pounds of ammonia with a minimum ullage of 2 percent at 120°F. The tank should have a radius of 8.82 inches. The factors accounted for (see Appendix A) in the sizing were (1) unusable propellant, (2) ullage, and (3) possible tank fabrication error. The actual storage capacity of the stainless steel tank delivered is 2796 cubic inches, which will store 56 pounds of ammonia. This capacity variation is attributable to tank fabrication problems that were not defined at the time of tank sizing. (See Appendix B.) Figure 1 is a photograph of the system.

Initial selection of the storage tank material was based on the requirement to provide adequate storage with flight-weight design. The initial material selected was titanium. Due to major problems associated with the fabrication of the titanium hemispheres (see Appendix B), a stainless steel tank was substituted. Though this tank provides some operational features of the titanium tank, it is significantly heavier and has a greater thermal capacity.

A feasibility stress analysis was made of the titanium tank and stainless steel regulation system. The results of this study indicated that the maximum stress intensity in the titanium tank would be 32.2 kilopound/inch^2(kpsi), which is 26.8 percent of minimum yield. The maximum secondary stress intensity is 99.1 kpsi, which is 82.6 percent of minimum yield. Although the latter could produce as much as a 10-percent change in the surface orientations between tank wall and regulation system mounting regions, we felt that, as there would be some angular mismatch in true assembly, the change would tend to be less severe than 10 percent.

At design conditions, the maximum primary stress in the regulation system plenum chamber is 3.66 kpsi, which is 12.2 percent of minimum yield. The maximum secondary stress is 84.8 kpsi, which would exceed minimum yield by 183.0 percent. This analytical result did not account for the end cap or the superstructure restraints. It also considered the chamber to be a 1/2 cylinder covered by a flat plate, though the top plate was slightly curved. Based on this analysis, some permanent set should have been produced during the hydro test, which was twice the pressure used in the analysis. Since no such effect was produced during testing, we feel that the simplified analysis was not realistic.

Dynamic stress conditions were found to be less severe than design conditions. Appendix C gives details of the preliminary stress analysis. Table I shows puchased components associated with the storage tank.

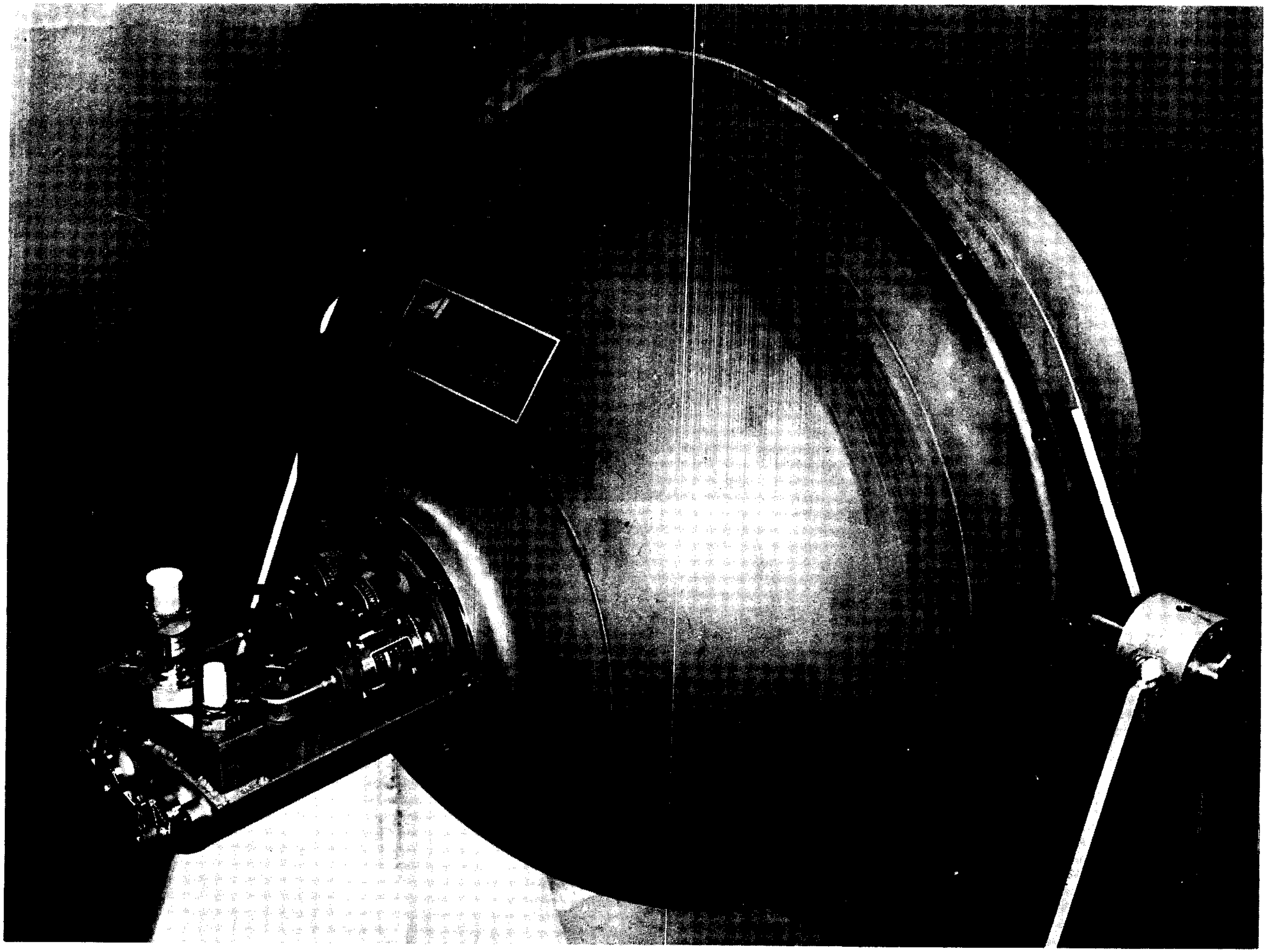

Figure 1. PHOTOGRAPH OF AMMONIA STORAGE AND FEED SYSTEM

TABLE I

PURCHASED COMPONENTS ASSOCIATED WITH THE STORAGE TANK

Component	Manufacturer	Model Number
Storage Tank	Elektron Standard, Inc.	---
Pressure Transducer	Micro Systems	1003-0046; 0-300 psia
Fill Valve	Pyronetics, Inc.	1176-46
Burst Disk Relief Valve	Carleton Controls, Corp.	1962-006-7

B. PRESSURE REGULATION AND PROPELLANT FEED

The function of the pressure regulation and propellant feed system is to provide gaseous ammonia on demand at a predetermined pressure. The specifications required that there be no more than 1-percent variation of the regulated pressure. The system is required to meet this specification with either gas or liquid input from the storage tank (must function under zero gravity conditions). A schematic of the storage and feed system is shown in Figure 2, and a photograph of the feed system is shown in Figure 3.

Referring to Figure 2, the principal elements of the regulation system are the pressure switch (PS), solenoid valve (V)--the system includes two of each for parallel redundancy: PS1, PS2, V1, and V2--and two chambers separated by an orifice. Flow from the storage tank to the preplenum chamber is controlled by the solenoid valve, which is actuated by the pressure switch.

Using a variable setting pressure switch (5—50 psid--manufacturer-stated dead band, 4.5 lb/in^2) set to shut off at 15 psia (worse control case), the preplenum pressure data shown in Figure 4 were obtained for gas and liquid flow from the storage tank over a range of supply tank pressures.

The effect of the dual chamber (preplenum and plenum) and the connecting orifice is shown by the plenum pressure fluctuation data in Figure 5. These data were taken at supply pressures of 115, 135, 155, and 230 psia, over a range of flow conditions. The worst-case pressure-regulation conditions are a regulated pressure of 15 psia, liquid flow from the storage tank at 155 psia, and a propellant use rate of 20×10^{-6} lb/sec. Under these conditions, the system controlled pressure to ± 1.01 percent. For all other flow or pressure conditions specified in the contract, it was possible to control regulated pressure to better than ± 1.0 percent.

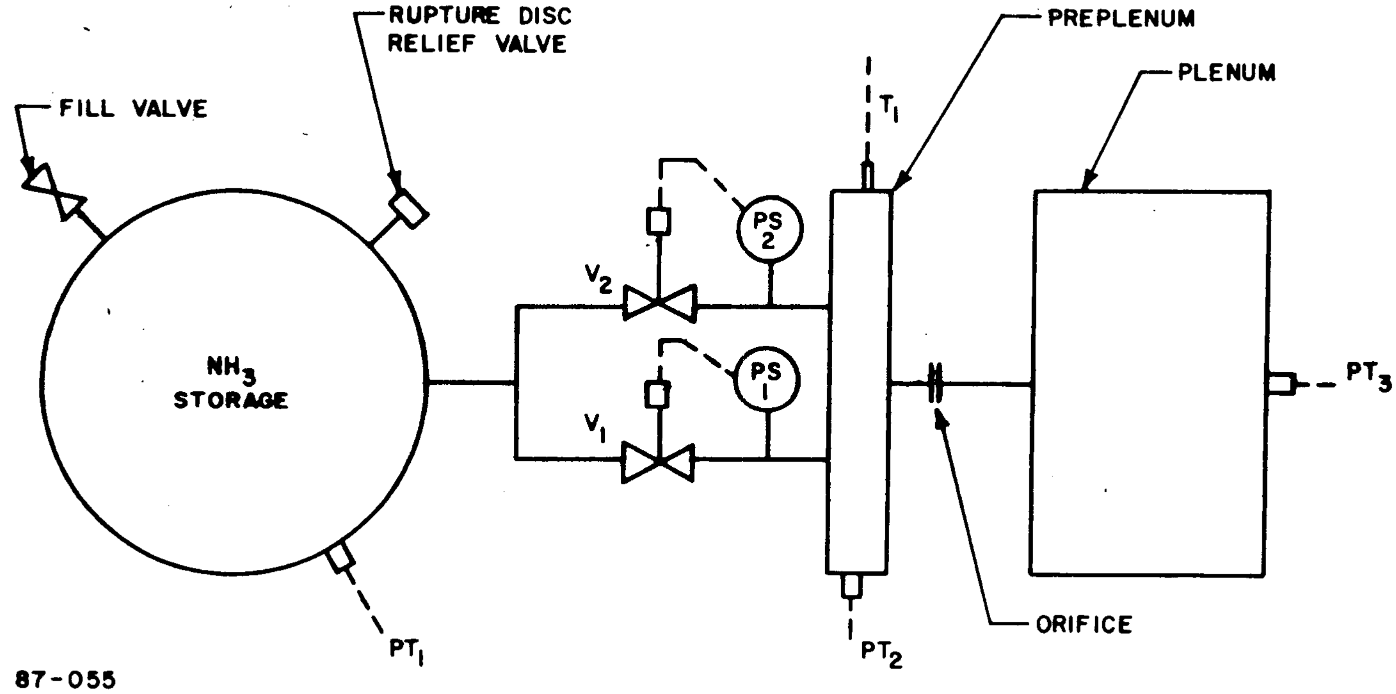

Figure 2. SCHEMATIC OF THE AMMONIA STORAGE AND FEED SYSTEM

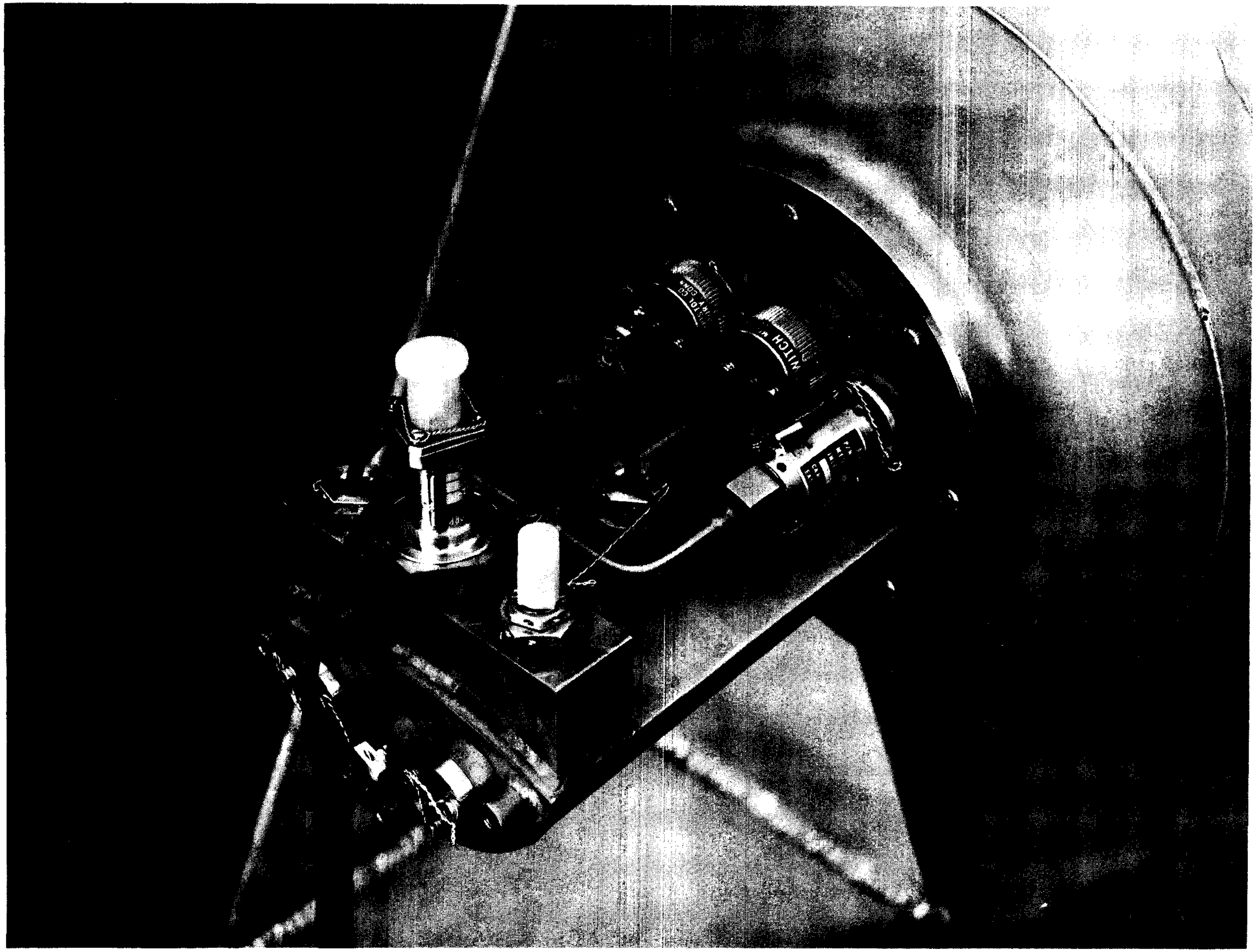

Figure 3. PHOTOGRAPH OF THE REGULATION AND PROPELLANT FEED SYSTEM

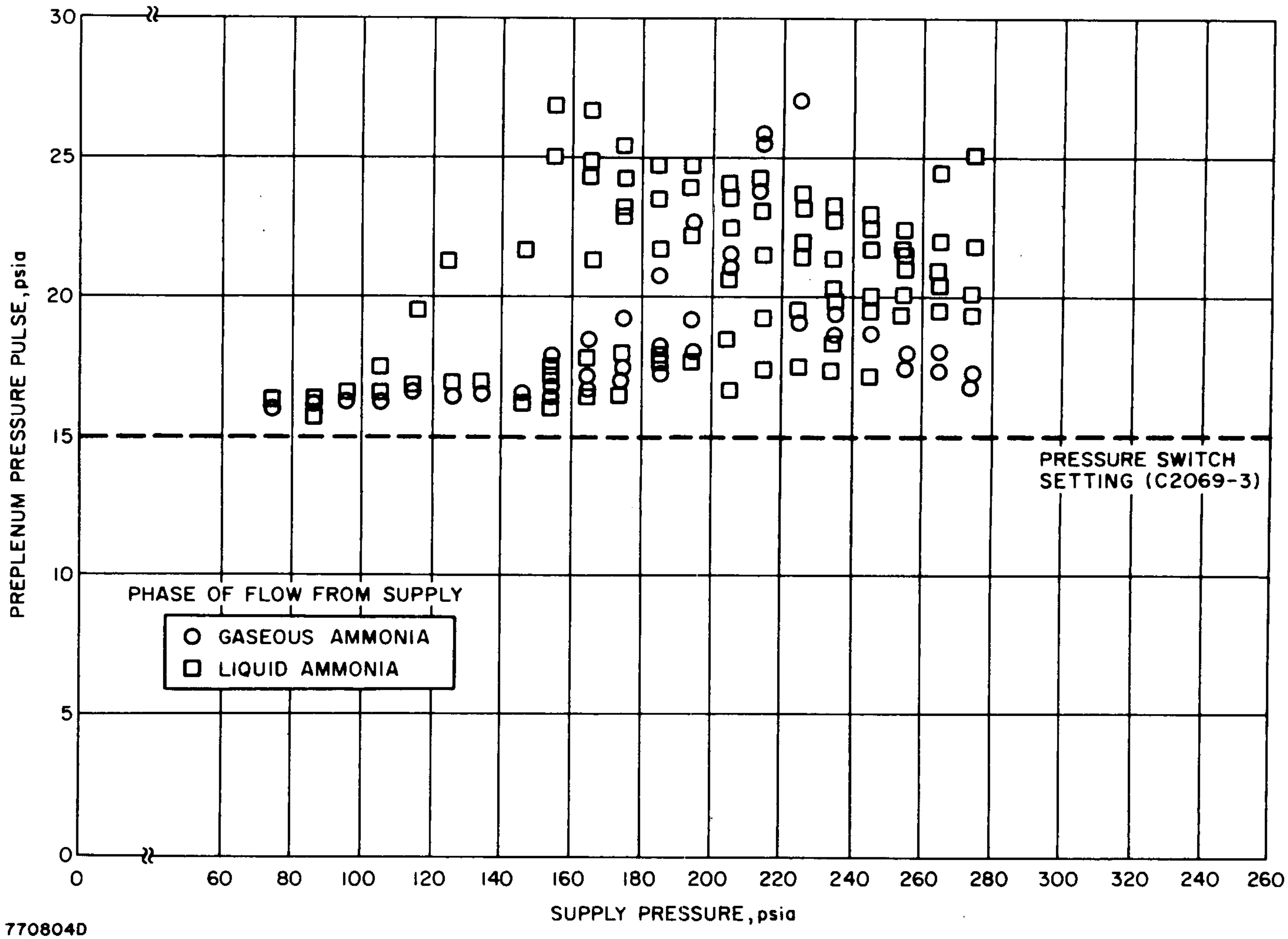

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Figure 4. PREPLENUM PRESSURE PULSE VERSUS PROPELLANT SUPPLY PRESSURE

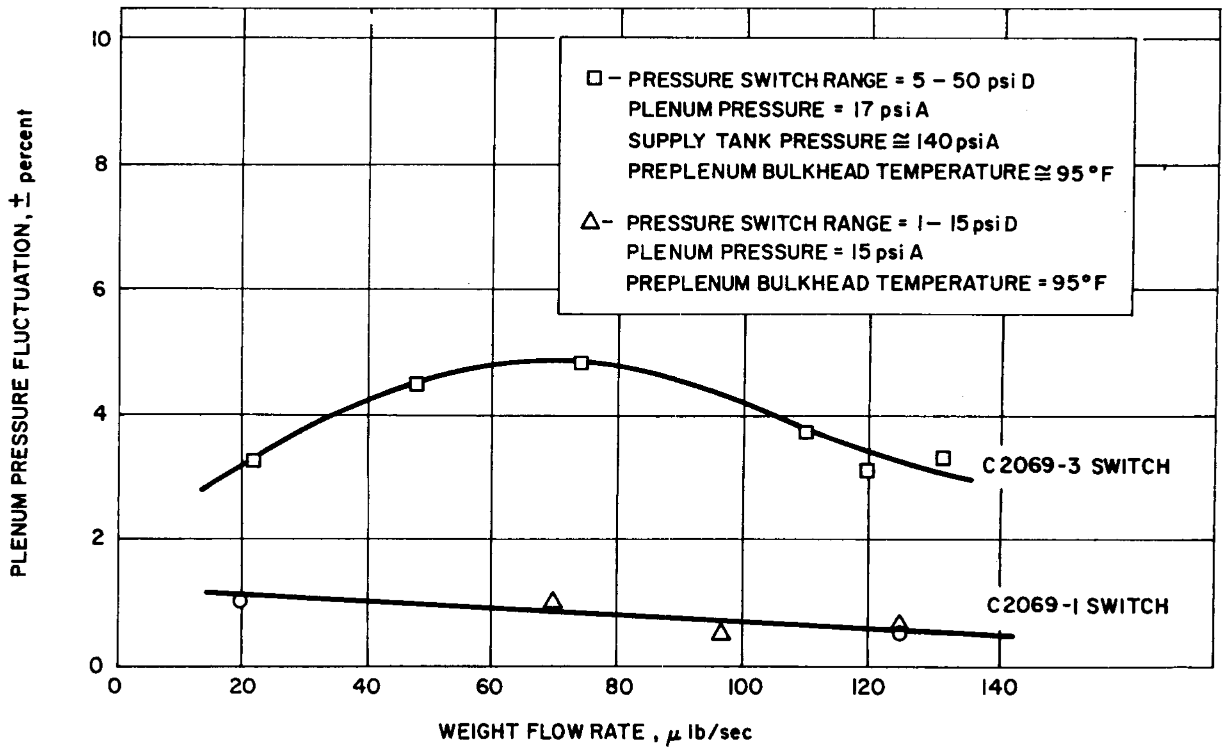

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Figure 5. REGULATED PLENUM PRESSURE VERSUS PROPELLANT USE RATE

The data of Figure 5 also demonstrate the effect on regulated pressure variation caused by the dead band of the pressure switch. The C2069-3 switch has a normal dead band of 4.5 psia. The C2069-1 switch has a dead band of 1.5 psia. To obtain ± 1.0-percent regulated pressure control, the pressure switch should be selected for its appropriate range of operation. This is indicated in Table II. (Note: There is in the system a basic tradeoff between valve cycling rate and pressure regulation variation--the better the control, the higher the valve cycling rate.)

TABLE II

PRESSURE SWITCH EFFECTS

Pressure Switch Model Number	Normal Operating Range (psid)	Normal Dead Band (lb/in^2)	Switch Setting (psia)	Maximum Preplenum Pressure Variation (percent)	Maximum Regulated Plenum Pressure Variation (percent)
C2069-1	1–15	1.5	15	53	1.01
C2069-3	5–50	4.5	15	80	4.86
C2069-3	5–50	4.5	50	22	0.95

Sections of recorded system performance data are shown in Figures 6, 7, and 8.

As indicated previously, the regulation system has been designed with a parallel redundancy of two regulating valves and two pressure switches. In this system, one switch is set at a pressure slightly below (0.5 lb/in^2) the other pressure switch. Therefore, should one valve-switch combination fail closed, the other would commence controlling when the plenum-preplenum pressure dropped the additional 0.5 lb/in^2 due to flow usage.

The components of the regulation and feed system are shown in Table III.

Early analytical predictions were nearly correct concerning the adequacy of the thermal capability of the system to provide the energy for ammonia vaporization. For the worst condition (i.e., a maximum flow rate, 248 x 10^{-6} lb/sec at 20°F for 208 seconds, with 3.30 pounds of propellant remaining in the storage tank), the system was able to maintain the specified control. During this test, the preplenum flange temperature was monitored. The data shown in Figure 9 were taken during two tests at station-keeping flow conditions with liquid ammonia entering the regulation system. The temperature of the propellant storage tank remained nearly constant for these runs. A third test was made withdrawing gaseous ammonia for 660 seconds. The total system temperature change was less than 1°F. This is also shown in Figure 9. (Note: The titanium tank with its thinner walls would have a larger temperature change, but, based on actual tankage weights,

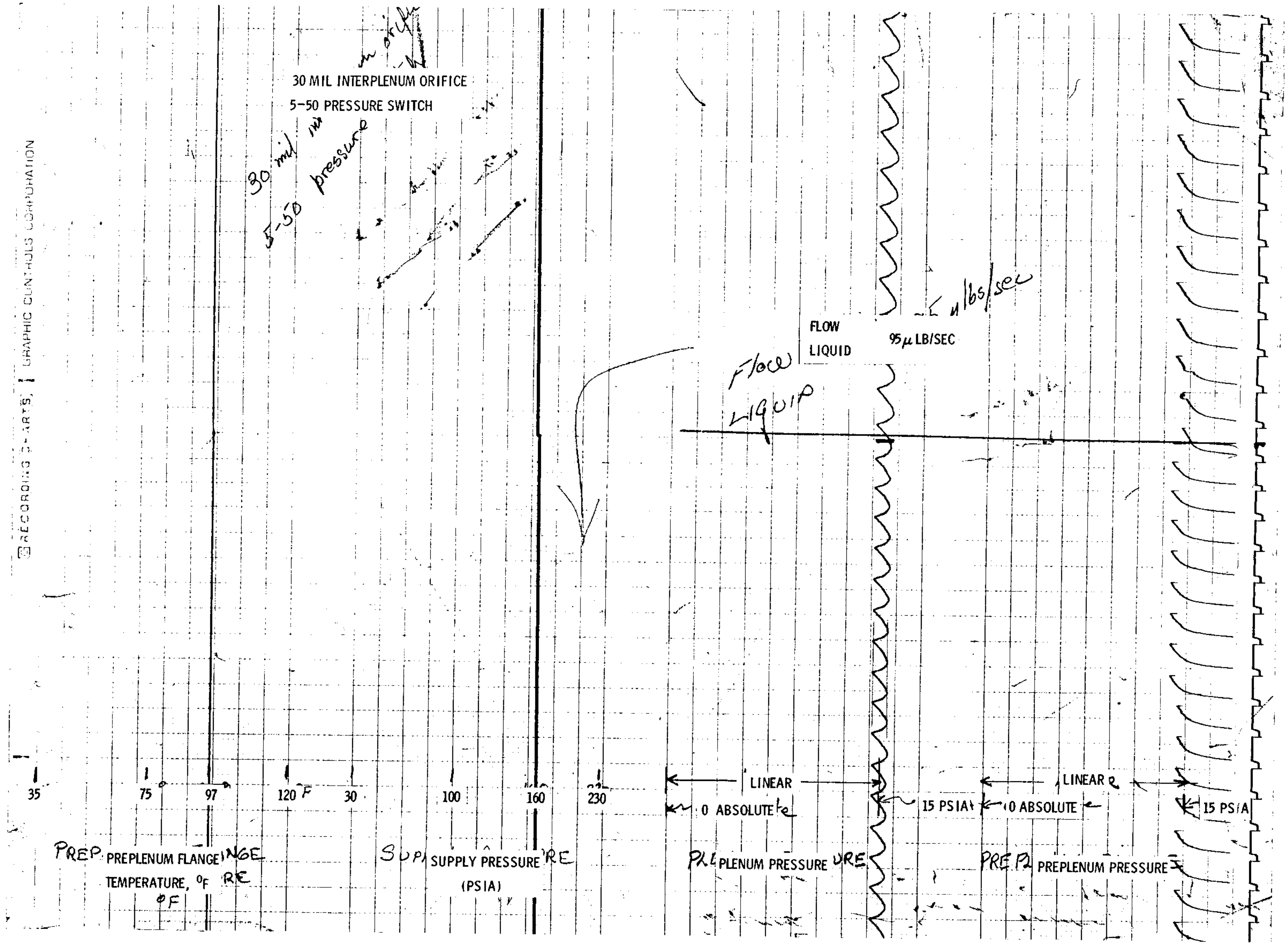

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Figure 6. SYSTEM PERFORMANCE DATA: LIQUID FLOW, 30-MIL ORIFICE, 5-50 PRESSURE SWITCH

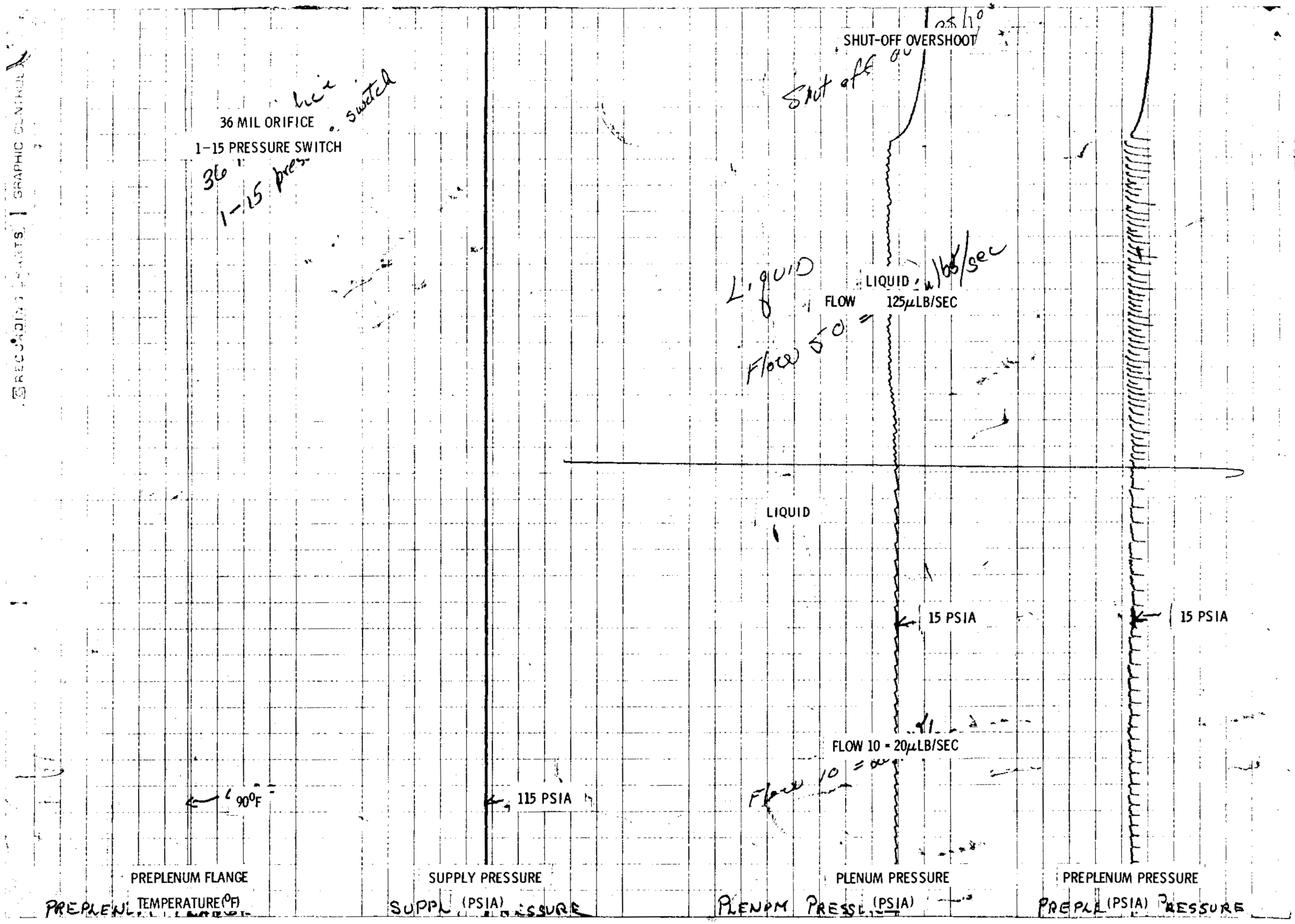

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Figure 7. SYSTEM PERFORMANCE DATA: LIQUID FLOW, 36-MIL ORIFICE, 1 - 15 PRESSURE SWITCH

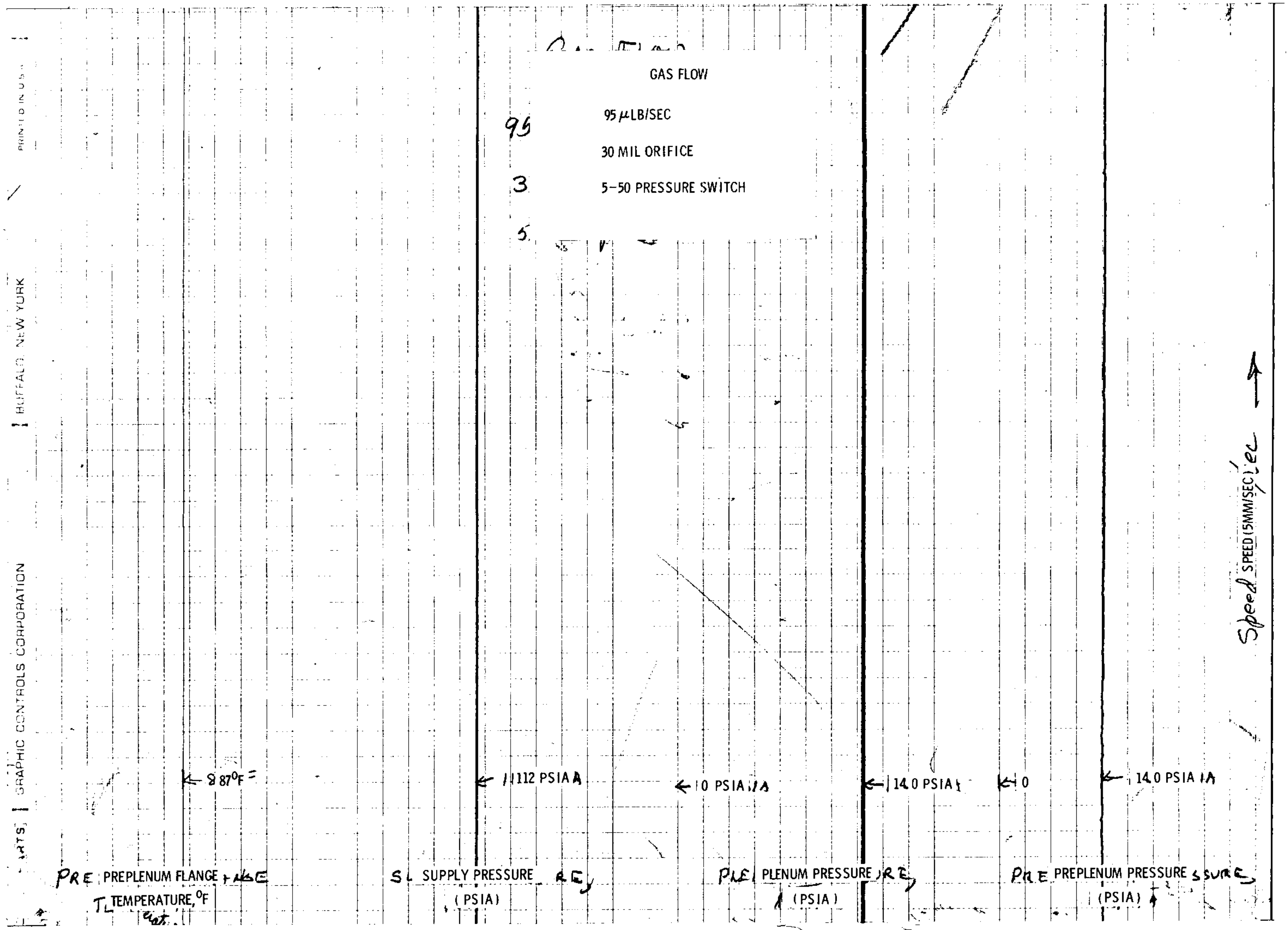

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Figure 8. SYSTEM PERFORMANCE DATA: GAS FLOW, 30-MIL ORIFICE, 5 – 50 PRESSURE SWITCH

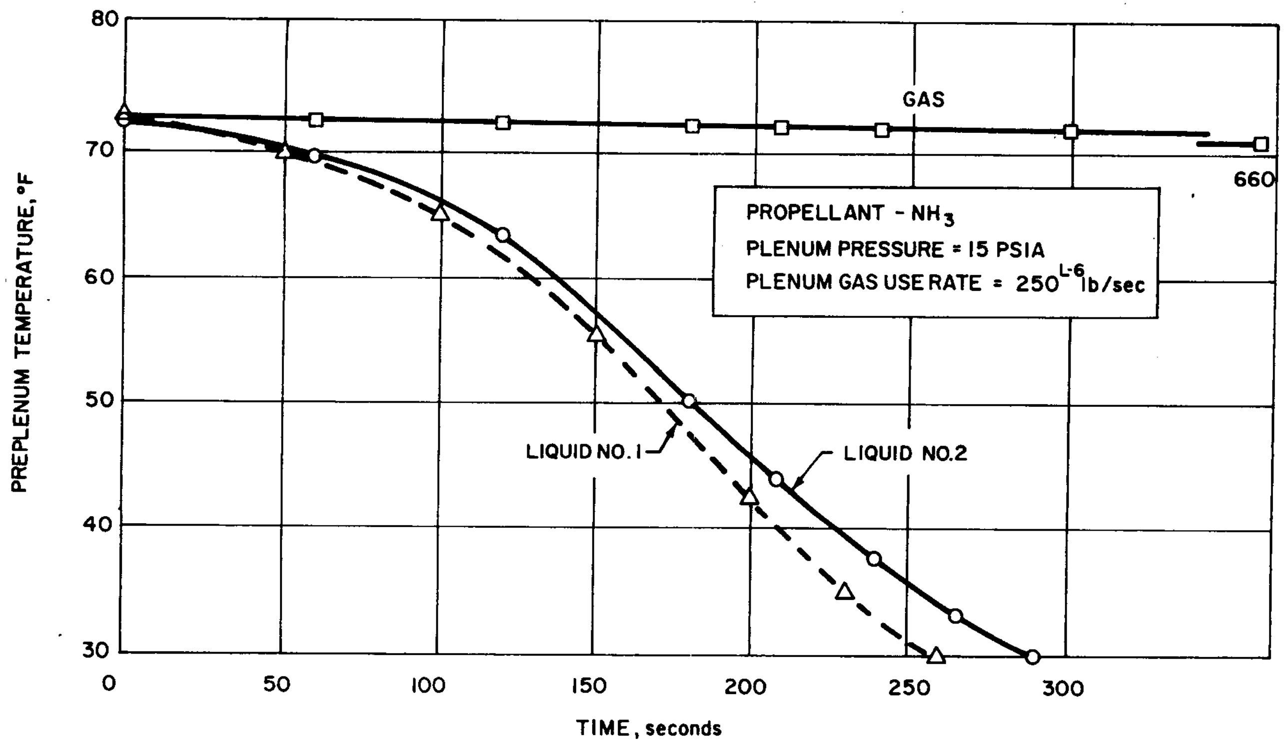

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Figure 9. PREPLENUM FLANGE THERMAL HISTORY FOR LIQUID AND GAS OPERATION

the temperature changes would be approximately those values calculated at the start of the program. Recorded data from these tests are shown in Figure 10.

TABLE III

PURCHASED COMPONENTS OF THE REGULATION AND FEED SYSTEM

Component	Manufacturer	Model Number
Regulating Valves	Carleton Control Corp.	1809-21
Pressure Switches	Bristol Co.	
0–15 psid		C2069-1
5–50 psid		C2069-3
Pressure Transducers 0–300 psia	Micro Systems, Inc.	
Thermistors	Fenwal Electronics, Inc.	K816

C. FLOW CONTROL ASSEMBLY

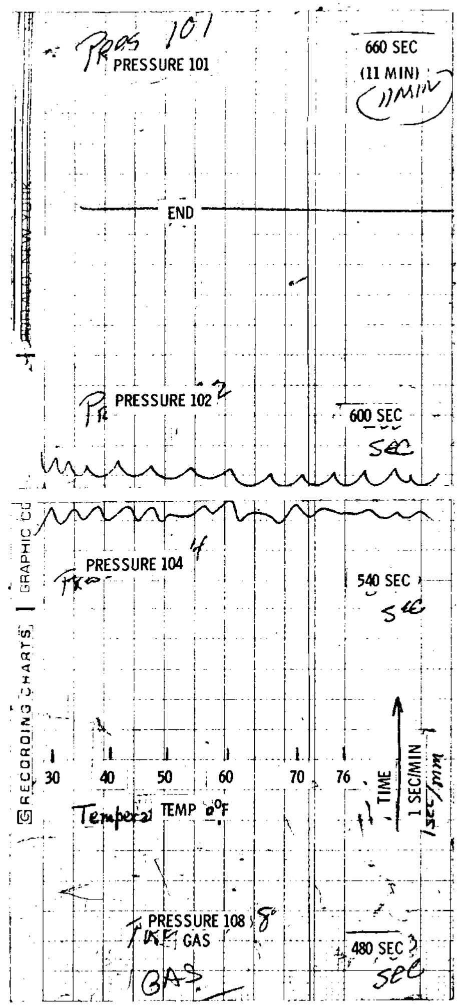
Table IV lists purchased components of the flow control assembly. Figure 11 is a photograph of the assembly.

TABLE IV

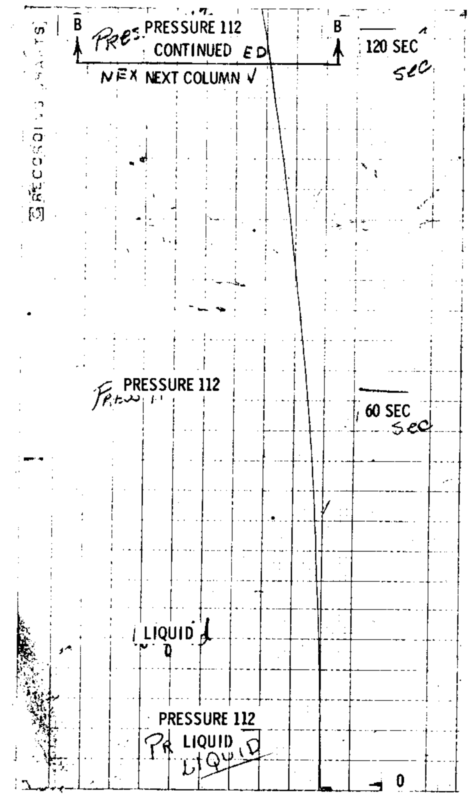
PURCHASED COMPONENTS OF THE FLOW CONTROL ASSEMBLY

Component	Manufacturer	Model Number
Flow Control Valve	Carleton Control Corp.	1809-20
Pressure Transducer	Micro Systems, Inc.	1003-0046 0–20 psia
Thermistor	Fenwal Electronics, Inc.	K816

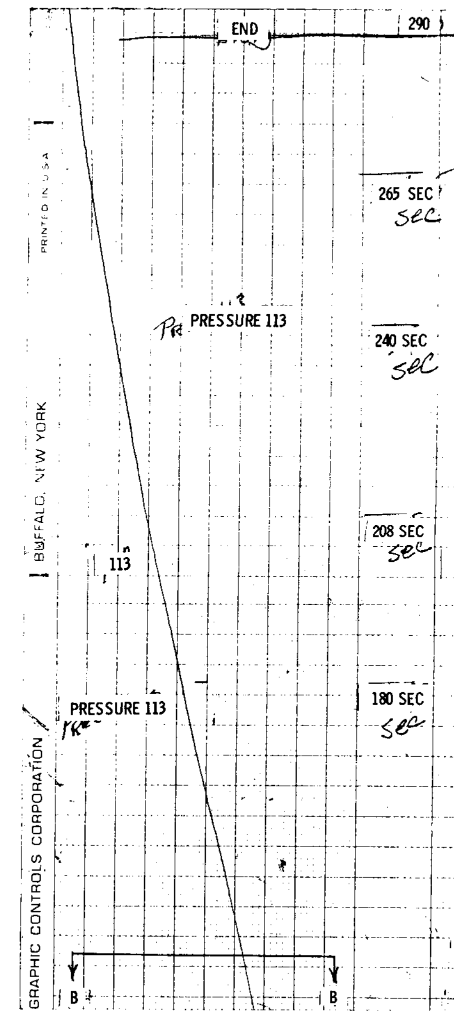
Efforts to obtain a satisfactory miniature solenoid valve or a latching solenoid have thus far proved unsuccessful. It will be noted in a following section that the obtaining of a standard ammonia usage solenoid valve has not been completely successful.



A. NH_3 GAS SUPPLY



B. NH_3 LIQUID SUPPLY



B. LIQUID TEST (CONTINUED)

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Figure 10. REPRESENTATIVE PREPLENUM THERMAL DATA

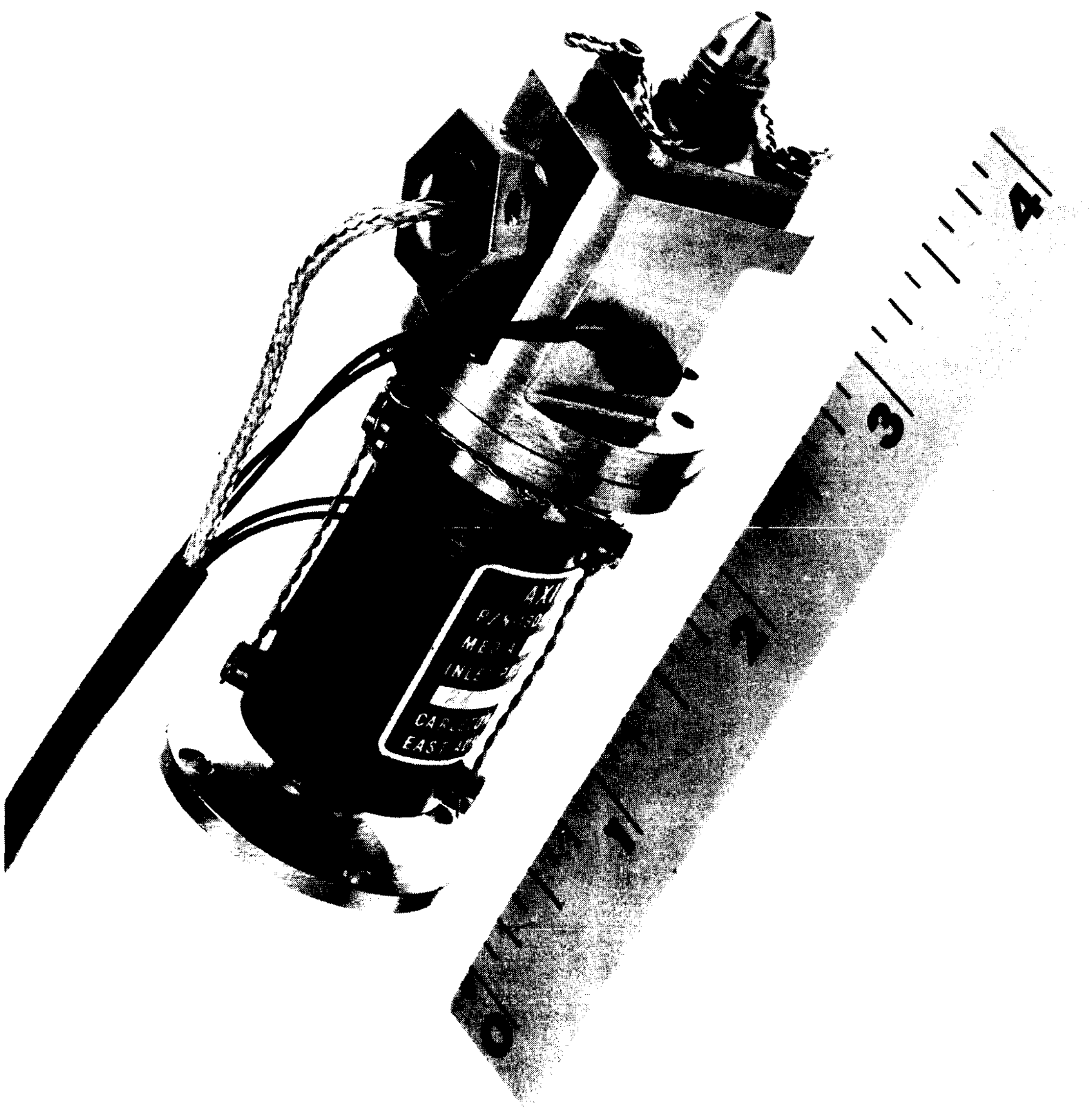

Figure 11. FLOW CONTROL ASSEMBLY

D. MASS FLOW INTEGRATOR

The mass flow integrator was developed to meet the requirements of providing a measurement of propellant mass or level when the system is in a test vacuum chamber. It was felt that could an instrument be developed that would also function in a space, zero-g environment, such an instrument would fulfill a need associated with several other types of propellant. Therefore, additional developments were started on a unit, conceived before this contract, that would use the pressure drop across the preplenum-plenum orifice for an analog measurement of the mass flow passing through the orifice. To eliminate the usual circuit drift, the integrated analog signal would be converted to a digital count, which would then be stored. The determination of propellant remaining in the fuel tank would be established by subtracting the propellant removed from the initial amount.

Coupling this concept to the actual operation of the ammonia regulation system involved considerably more activity than anticipated. In a test run, however, with a laboratory breadboard model of the mass flow integrator and an ammonia feed system, the integrator operated completely satisfactorily. Both with liquid- and gaseous-ammonia flow being introduced into the regulation system, the integrator counted the mass flow with an accuracy of better than 97 percent. Temperature effects could account for most of the counting error.

Previous tests of the integrator with air flow were compared to measurements made from flow orifice calculations. Several runs at given mass flow rates and at three plenum pressures were made and the results averaged. In general, the peak-to-peak count variation for the seven count intervals was very small, ammounting to perhaps 1.5 percent. (Details are provided in Appendix D.) The count accuracy at each of the three plenum pressures showed deviations of less than 1 percent for flow rates from about 50 to 150 μ lb/sec. Finally, the cumulative error when both varying plenum pressure and mass flow rate was about 7 percent maximum. This cumulative error can be reduced, as the runs were made with a preplenum transducer sensitivity having a positive output at zero pressure. The error introduced by this would tend to nulify the cumulative error.

III. FUELING PROCEDURE AND EQUIPMENT

Three factors are here considered for the fueling operation: (1) the purity of the propellant, (2) the cleanness of the fueling equipment (particularly the connecting points), and (3) the amount of propellant transferred. The actual fueling procedure is outlined below, together with several general comments about the operation.

1) Evacuate storage tank to at least 1 torr, and maintain vacuum pumping for 1 hour. Then, close fill valve and turn off and disconnect pump. This allows sufficient and accurate fill of ammonia and reduces contamination of the tankage system.

2) Use a stainless steel pressure container for portage of the ammonia to the storage tank. This container should be equipped with a 1/4-inch Kel-F seat manual valve (Whitey type) and a very short length of 1/8-inch stainless-steel tubing, having an AN-type fitting for attachment to the fill valve. The container should be ultra-sonically cleaned (no chlorine), evacuated, weighed, and then filled with high-purity ammonia (careful not to overfill). The container is then reweighed and connected, and the ammonia transferred to tank. The container is again weighed to determine the amount transferred. (Over filling can occur should the density of the liquid ammonia increase during the transfer process, and should the container be nearly filled with higher density ammonia: when the ammonia is warmed, the density decreases and extreme pressures can be obtained.)

3) After connecting to the fill valve, the fill valve is opened with a 9/16-inch wrench by turning the thin concentric nut counter-clockwise until it again begins to indicate resistance to turning. (Before operating system, operator should be familiar with the valve. Allow a small amount of ammonia to be transferred into the storage tank. Close the manual transfer line valve and then the storage-tank fill valve. Allow tank to arrive at ambient conditions. This will accomplish the following:

a) Control the rate of pressure rise in the storage tank.

b) Allow for leak checking in system.

c) Allow subsequent liquid transfer with a minimum of local temperature drop. (When liquid flashes into the evacuated tank and impinges on the wall opposite the fill tube, a large local temperature drop will occur.)

4) Re-open storage-tank fill valve, and then open manual transfer line valve to continue fuel transfer process. Fueling now should be accomplished as quickly as possible in order to keep the time available for liquid expansion short.

The filling procedure is accelerated if the fill container is higher than the exit tube. To further accelerate the transfer, a heat gun may be directed over the fill container; be certain, however, that both the fill valve and hand valve are open before applying heat. Another technique is to chill the propellant storage tank. This may be done with ice packs, ice bath, or cooling coils.

Usually, the completion of the transfer can be determined by the end of the sound of running liquid. The fill valve is then closed with 5 inch-pounds of torque, the hand valve shut off, and the pressure in the transfer line released slowly as the connecting fittings are loosened. This transfer line should be kept extremely short downstream of the hand valve.) This procedure is repeated until sufficient propellant has been transferred.

The total tank volume has been measured at 2796 inches.[3] Therefore, the maximum amount of NH_3 that should ever be loaded is 56.0 pounds.

IV. REMAINING PROBLEM AREAS

In general, the system delivered meets all of the specifications of the contract. Several significant problem areas remain, however. Principal of these are (1) the availability of a reliable ammonia-usage solenoid valve, (2) fabrication of a thin-wall titanium storage tank, (3) evaluation of materials compatibility with a realistic purity ammonia, and (4) fabrication and testing of a flight-qualified mass flow integrator.

A. SOLENOID VALVE

The most consistent problem in the program has been the solenoid valves. This problem stems from the fact that existing qualified valves have been modified and adapted to a new application. Before the start of this program, Avco had done a survey of available valves for use on ammonia auxiliary propulsion systems. The findings showed that the desired valve was not available. Based on inputs from Avco, Carleton Controls adapted an existing valve for low-flow ammonia. The designation of this new valve series was 1809.

Initial laboratory use of this valve demonstrated a problem of flow direction of liquid ammonia and valve bellows. Following correction of this problem, the valve was put back into life testing and system use. During life tests, several of the valves successfully passed (leak rates less than 6×10^{-5} STD cm^3/sec of He) hundreds of thousands of cycles. It was later observed, however, that this same valve could fail after a few (10 to 1000) cycles when exposed to high purity ammonia. The problem was diagnosed to be the chemical bonding agent used to hold the valve seat in place. Solution of this problem followed development and test of several methods of mechanically holding the seat in place.

Later, another major valve problem was discovered. Throughout the period of use of the 1809 valve, a problem diagnosed as valve adjustment was observed. Valves were assembled and checked at Carleton Controls and shipped to Avco. Subsequent testing of the valves at Avco indicated valve sticking or high leak rates. Additional valve adjustment would correct the problem. The Massachusetts Institute of Technology, Lincoln Laboratory, reported the same problem. Several times during the environmental testing of this valve, changes of adjustment were observed. During August, the adjustment problem was determined to be the result of the design of the valve case and the small case set screws. (A partial turn of one of the small set screws could change the valve adjustment from fully closed to leaking.) A further modification to the valve apparently corrected this problem; during a later vibration test, however, a modified valve was observed to be stuck closed and would not respond when a voltage was applied to open the valve. (The valve functioned properly after being tapped.) The environmental testing of the valve has also demonstrated that this valve would dribble during vibration loading.

Carleton Controls was given a full report of the valve test results. They were also informed that both Avco and Goddard (W. Lund and D. Suddeth) considered the valve unsatisfactory. The following recommendations were also made:

1) Valve housing should be either welded together or made in one part (permanent assembly).

2) The necessity for valve adjustment should be completely eliminated.

3) The requirements for eight O-rings within the valve should be eliminated. Welding following assembly was recommended.

4) Should the valve require mechanical assembly, the correctness of the assemblies must be capable of verification by an external positive test.

5) The valve must remain closed when subjected to a launch environment.

6) The valve must open when energized with an equivalent solenoid force of 75 g.

7) Reliability is more important than minimum power use.

These recommendations, together with the performance and quality assurance specifications, were also given to other valve manufacturers. New valve conceptual designs were received from three companies. These designs were reviewed with NASA/Goddard, NASA/Lewis, and M.I.T. Lincoln Laboratory. The review comments were in turn relayed to the respective manufacturers. Prototype valves for evaluation testing were later received from two companies. During the initial testing, one of the valves failed leak-rate tests. The valve was returned to the vendor, who diagnosed the problem as contamination inside the valve. This valve was built with filters at both ends. Four valves from the second company were put into preliminary cycling with ammonia flow before leak testing. Due to external leakage of ammonia through the valve housing, all four valves had to be returned to the vendor. This leakage was the result of the ammonia reacting with the solder holding the housing together. (The vendor said that the reason he had not welded the valve together was to allow such repairs or modifications to be made that should be required as a result of the evaluation testing.)

B. FABRICATION OF A THIN-WALL TITANIUM TANK

In theory, the most economical method for fabrication of a thin-wall titanium hemisphere would be by spinning. Tank assembly could then be completed by welding. As indicated in a previous section and in Appendix B, however, significant problems have been observed using this process. An Avco funded program with Elektron Standard, Inc., and Spincraft of Milwaukee is attempting to fabricate the desired tank by spinning and electron-beam welding. On this program, the spinning procedures outlined in Appendix B were used to spin acceptable hemispheres. At the time of the preparation of this report the hemispheres were in the process of being assembled with the mounting flanges by electron beam welding.

C. MATERIALS COMPATIBILITY WITH AMMONIA PROPELLANTS

Materials handbooks indicate a great number of materials, including aluminum and copper, are compatible with chemically pure ammonia. In actual usage, however, it has been observed that "high-purity" ammonia obtained from suppliers is not pure. There is, for example, always some quantity of water. (Avco always processed this "high-purity" ammonia through several additional distillations and filterings.) There is also some degree of contamination carried by the contact surface of the respective material. Consequently, in actual usage, the compatibility guides for pure ammonia cannot be followed.

A program should be established to evaluate the compatibility factor for temperature ranges of interest for pure ammonia and propellants containing ammonia, such as many of the subliming solids.

D. MASS FLOW INTEGRATOR

This instrument shows promise not only for use in ammonia systems, but also with all other liquid or solid propellants that are converted to a gas before expulsion from the thrust nozzle. We believe that, with the addition of an instrument refinement of a temperature compensator, measurement accuracies of greater than 95 percent can be obtained with a light-weight (less than 0.25 pound) low-power (less than 1 watt) flight unit.

APPENDIX A

SIZING OF THE PROPELLANT STORAGE TANK

The volume of the propellant tank, accounting for the preplenum boss is

$$V = 4/3\ \pi R^3 - \pi h^2 (R - h)\ ,$$

where R is the tank radius, and h is the height of the spherical segment of tank removed by the preplenum. Determination of volume is based on the tankage and ullage requirements. These requirements are reviewed in the following section.

A. UNUSABLE PROPELLANT

Propellant remaining in the tank, preplenum, and plenum when the pressure in these volumes has reached the assigned minimum lock-up pressure constitutes the depleted condition. The maximum specified lock-up pressure is 75 psia.

The following are the approximate volumes involved:

Tank	=	2891.0 in^3
Preplenum	=	0.5 in^3
Plenum	=	40.0 in^3
Total System Volume	=	2931.5 in^3

The amount of gas contained in the system at 75 psia and 20°F (the conditions for maximum residual mass) can be determined from the universal equation:

$$M = \frac{mPV}{RT}\ ,$$

where

M = weight, pounds

m = molecular weight = 17.03 pounds for NH_3

P = absolute pressure = 75 x 144 (lb/ft^2)

V = volume 2932/1728 (ft^3)

R = universal constant, NH_3 = 1537.8 (ft-lb/°R)

T = temperature = 460 + 20 = 480 (°R)

M = 0.4228 pound.

Propellant lost from the system during a 3-year mission has been estimated on the basis of measurements taken on a previous laboratory ammonia storage and feed system. This is felt to be a reasonable approximation, as the number and length of tank welds, number of valves, types of valves, etc., are similar. This value is 0.100 pound.

B. ULLAGE

All criteria for tank ullage in a flight system are difficult to establish. Ullage for this system has been established by providing a full tankage at 130°F, ammonia density to 34.67 lb/ft^3, rather than the specified maximum operational temperature of 120°F. This allows approximately 2 percent ullage at 120°F. (Note: The system is additionally protected by a rupture disk and relief valve.)

C. VOLUME SENSITIVITY TO FABRICATION ERROR

The maximum anticipated fabrication error is 2 mils. For a tank radius of 8.816 inches, the percentage of volume change per mil error is 0.034 percent. Therefore, a 0.07-percent volume error is associated with a 2-mil error.

From the preceding factors and the specified 57.00 pounds of usable propellant, a propellant load of 57.523 pounds is established. From the value of 34.67 lb/ft^3 (value associated with 130°F), the required tank volume is 1.659 ft^3. Following an iterative process, the tank radius was established as 8.816 inches.

APPENDIX B

PROBLEMS ASSOCIATED WITH FABRICATION OF A TITANIUM TANK

Elektron Standard, Inc., of South Windsor, Connecticut, the firm selected by Avco to have responsibility for fabrication of the titanium tank, surveyed 10 titanium metal-forming companies before selecting Eastern Metals, of East Hartford, Connecticut, for fabrication of the titanium hemispheres. These hemispheres were to be subsequently assembled by electron beam welding by Elektron Standard.

The first-finished spun hemispheres exhibited many stress lines and multiple cracking around the periphery, 3 inches from the edge. Subsequent spinning attempts by Eastern, using extra stress relieving operations (950°F for 8 hours) between spinnings, showed little improvement in the final hemispheres.

The final approach used by Eastern was to start the operation with a thicker material and use contour machining following the spinning operation in order to obtain the final configuration. To gain experience with this approach, the technique was used to fabricate the stainless steel backup tank ordered by Avco. The results were successful, although the tolerances on the internal dimensions of the tank were not maintained. Subsequent work on the titanium hemispheres produced better results than obtained previously, but they were still far from satisfactory, and the parts had to be rejected.

The spinning procedure used by Eastern was to heat the material with torches to 1100° to 1300°F for each break. Between breaks, the material was annealed.

In a subsequent attempt, Eastern gave the material a 2 to 4 micro-finish before starting the spinning. The final product of this effort was equally unsatisfactory.

The following comments and recommendations are based on an Avco study of the spinning procedures used for the titanium.

1) The titanium should be heated more uniformly than can be done with torches. Radiant heating lamps are suggested.

2) The torch heating likely produces hydrogen embrittlement of material.

3) Final rounding of the material could be achieved with inside machining. This would eliminate the final break of the spinning procedure. (Most of the part failures were observed to occur in the region of the final break.)

APPENDIX C

PRELIMINARY STRESS ANALYSIS OF A TITANIUM PROPELLANT STORAGE TANK AND A STAINLESS STEEL REGULATION SYSTEM PLENUM CHAMBER

A. STATIC ANALYSIS OF THE PROPELLANT STORAGE TANK

1. Design Conditions

The tank contains 57 pounds of saturated ammonia at a maximum temperature of 120°F, which corresponds to 288 psia. External pressure is zero. The tank is titanium, and the plenum chamber attached to the tank is stainless steel. Thus, there is a differential thermal expansion at 120°F, since assembly takes place at room temperature. The tank is supported at the bolt circle. Support reaction is assumed negligible under static design conditions. Material properties are the following:

	Titanium	Stainless
Yield strength	120 klb/in^2	30 klb/in^2
Ultimate strength	130 klb/in^2	75 klb/in^2
Coefficient of expansion	4.8×10^{-6}	9.6×10^{-6}
Young's modulus	16, 000 klb/in^2	28, 000 klb/in^2
Density	0. 16 lb/in^3	0. 29 lb/in^3

Ti-6Al-4V properties from MIL-HDBK-5, annealed condition 304 SST properties from U.S. Steel Data Sheet, annealed condition.

2. Static Analysis Model of Storage Tank

The titanium tank has been divided into axisymmetric shells and rings for analysis by a previously developed computer program (Program 1888). These regions are shown in Figure C-1 and described in the following paragraphs.

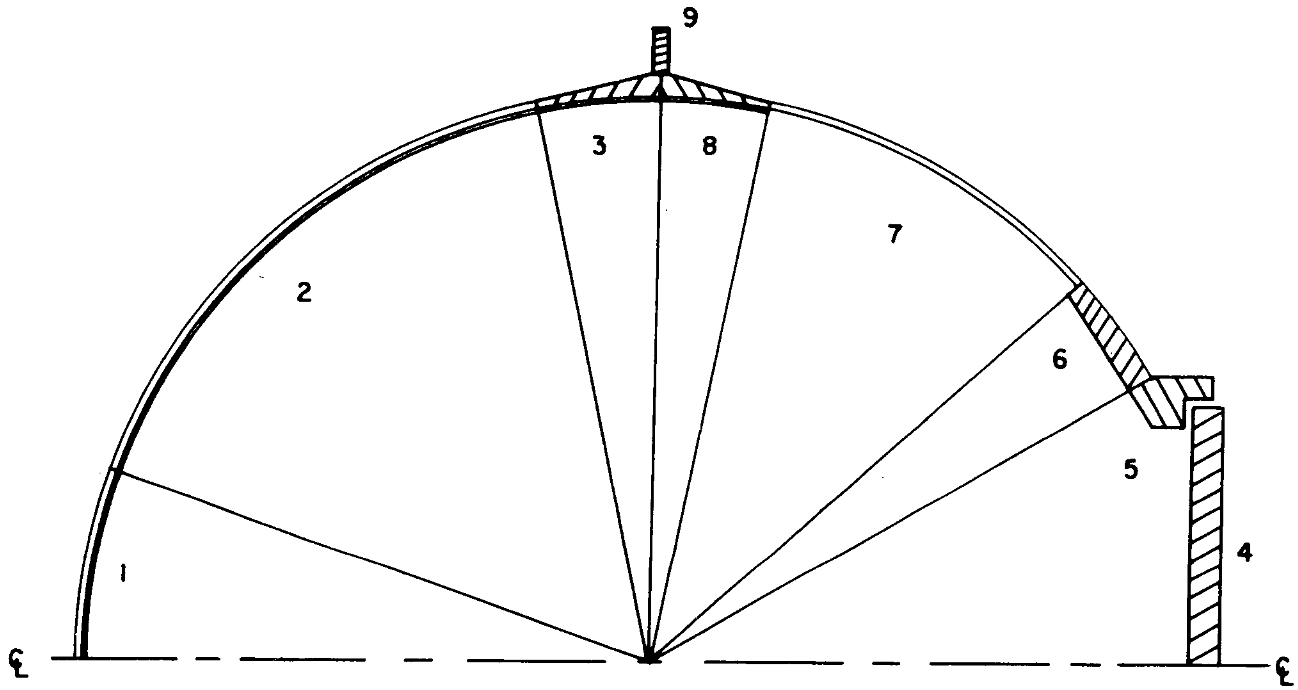

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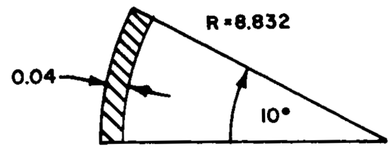
Figure C-1. ANALYTICAL AXISYMMETRIC SHELLS AND RINGS OF THE TITANIUM STORAGE TANK

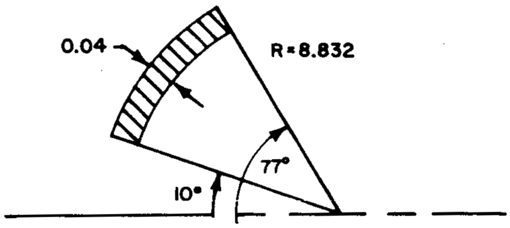
Region 1



Material: Titanium
Pressure: 288 lb/in^2
Temperature: + 50 degrees

This short region is employed to improve accuracy at the pole point. Shell thickness is the minimum.

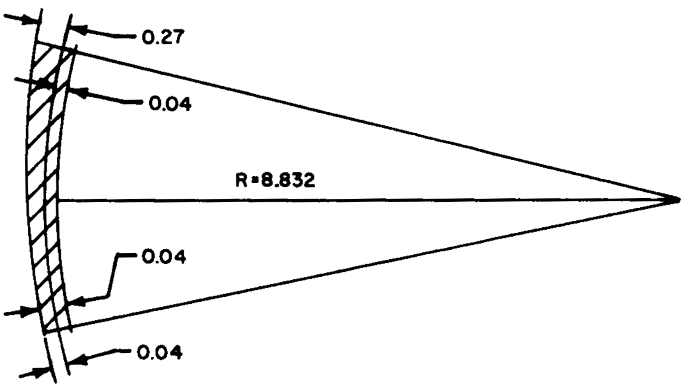
Region 2



Material: Titanium
Pressure: 288 lb/in^2
Temperature: + 50 degrees

This region completes the constant thickness spherical end.

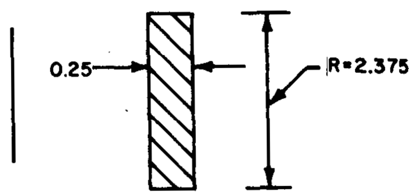
Region 3



Material: Titanium
Pressure: 288 lb/in^2
Temperature: + 50 degrees

The reference surface is taken as a continuation of the spherical surface of Region 2 in order to account for the thickness discontinuity. The thickness variation is assumed linear between the region end points.

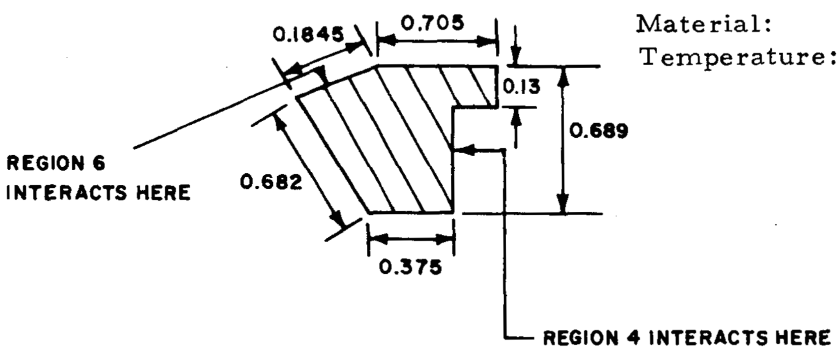
Region 4



Material: 304 SST
Pressure: 288 lb/in^2
Temperature: + 50 degrees

This region simulates the plenum chamber end plate; 2.375 is the bolt circle radius. 0.25-inch thickness is a compromise with the penetrations and other areas where material is removed.

Region 5



Material: Titanium
Temperature: + 50 degrees

Region 5 is a ring made up from the material left between 4 and 6.

Cross section area = 0.4 in^2

Cross section inertia = 0.0125 in^4

Radius to c.g. = 2.4 inches

Axial location of interaction points with adjacent regions:

Region 4: 0.2 inch to right of c.g.

Region 6: 0.394 inch to left of c.g.

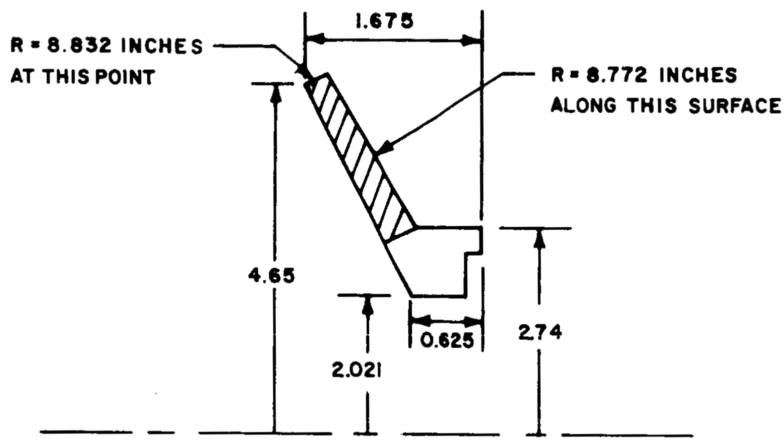
Net pressure loads on ring:

171.1 lb/in, radial

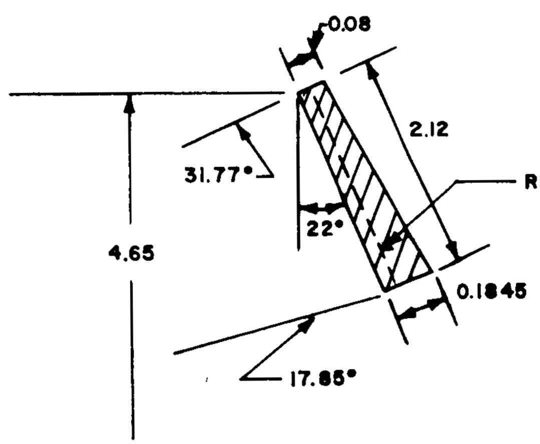
220.8/2.4 lb/in, axial

55.96 in lb/in, moment

Region 6



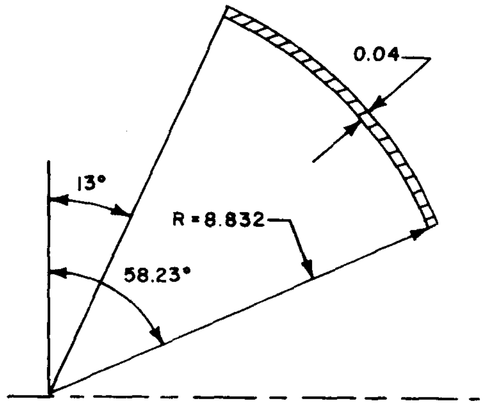
Starting with the dimensions shown above, we computed the following Region 6 dimensions.



MATERIAL : TITANIUM
PRESSURE : 288 lb/in^2
TEMPERATURE : + 50 degrees

The thickness variation is assumed linear between the region end points.

Region 7



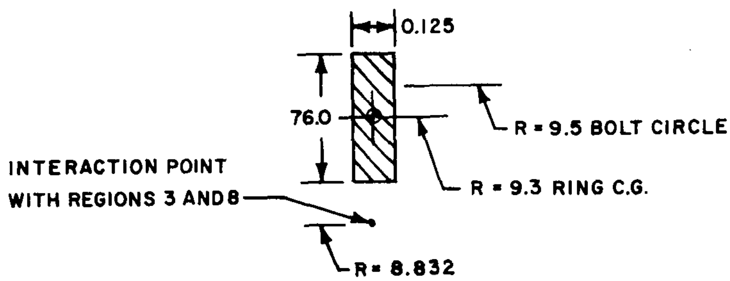
Material:	Titanium
Pressure:	288 lb/in^2
Temperature:	+ 50 degrees

Region 8

Region 8 is a mirror image of Region 3.

Region 9

Material:	Titanium
Temperature:	+ 50 degrees



This region is a ring cross section: area = 0.12 in.

cross section: inertia = 1.5×10^{-4} in^4

No net pressure load on this region. Interaction points are on vertical line through c. g.

3. Results of Computer Analysis at Design Conditions

The preceding model was run on Avco Program 1888, Case 2.0, Memo A-3246, June 3, 1966. The H card reads as follows:

H Titanium Ammonia Tank, 288 lb/in^2/120 degrees

The equivalent stress is plotted in Figure C-2; the equivalent stress or stress intensity is given by

$$\sigma_e = \left(\sigma_f^2 + \sigma_\theta^2 - \sigma_f \sigma_\theta \right)^{1/2}$$

Both primary (membrane) stress intensity and secondary (membrane plus bending) stress intensity are plotted. Maximum values are the following:

Primary = 32.2 klb/in^2, middle Region 7

Secondary = 99.1 klb/in^2, Region 7 and 6

B. STATIC ANALYSIS MODEL OF PLENUM CHAMBER

1. Design Conditions

The plenum chamber contains gaseous ammonia. The maximum operating pressure is 75 lb/in^2. The chamber is fabricated with 304 stainless steel.

2. Static Analysis Model

Because of symmetry, only half of the cross section of the plenum chamber was analyzed. The chamber was treated as a frame structure and analyzed by a previously developed computer program (STRESS Program). The cross section of the chamber and frame model used for the analysis are show in Figure C-3.

3. Results of Computer Analysis at Design Conditions

The model was evaluated on Avco Program STRESS, Memo A-3259, May 28, 1966, "Structure Plenum." The results are shown in Table C-I.

Maximum primary stress was 3660 lb/in^2. Maximum secondary stress was 84.8 klb/in^2. The axial stress is on the order of one half the primary stresses computed above. Furthermore, any axial bending will reduce the secondary stress in the frames. Therefore, the stress intensities will not exceed the stress components presented in Table C-I. Near the chamber ends, the maximum axial secondary stress, due to the edge restraint, will be less than twice the maximum primary stress above, i.e., less than 7.3 klb/in^2.

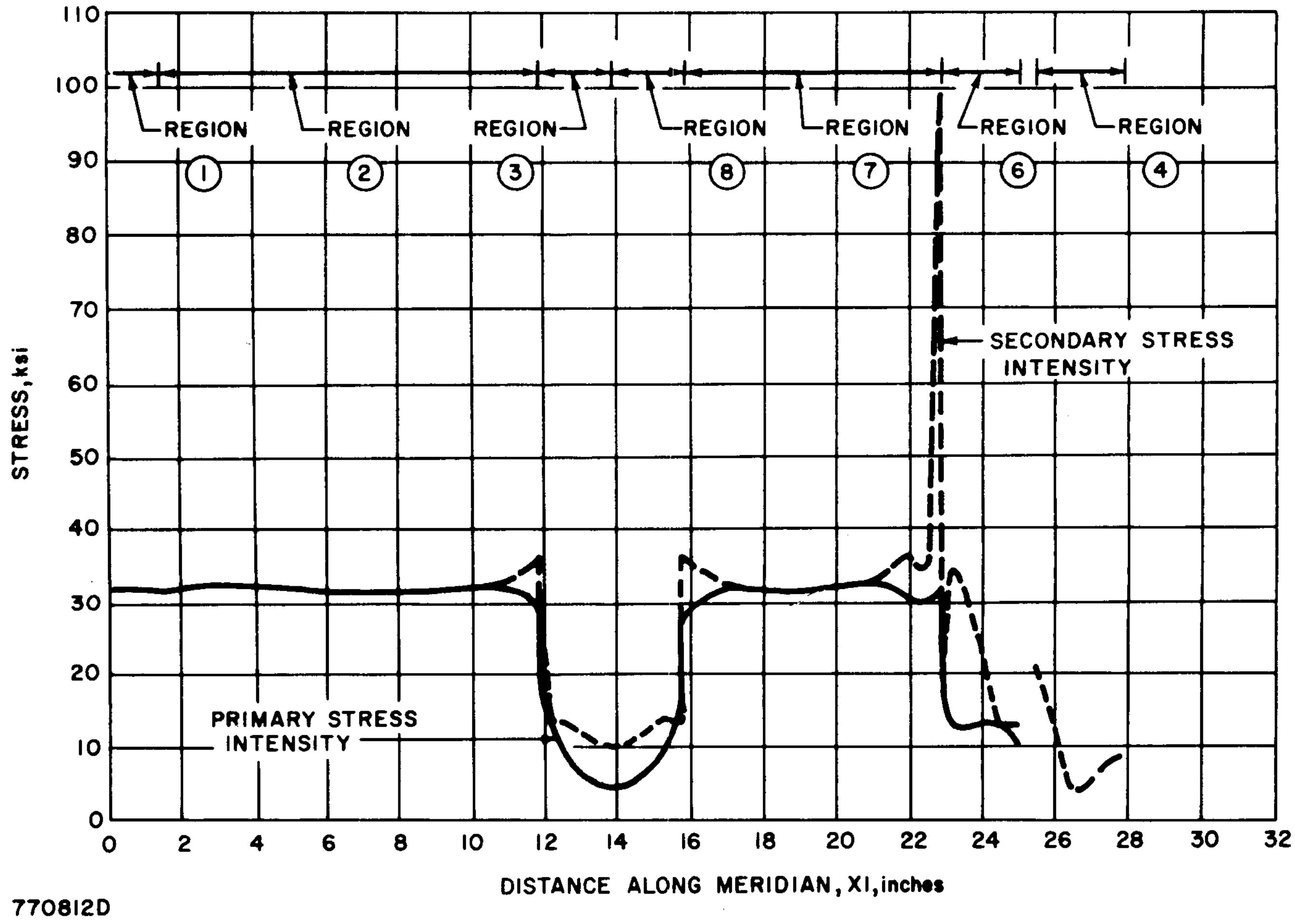


Figure C-2. AMMONIA STORAGE TANK STRESS INTENSITY AT DESIGN CONDITION
288 LB/IN^2, 120°F

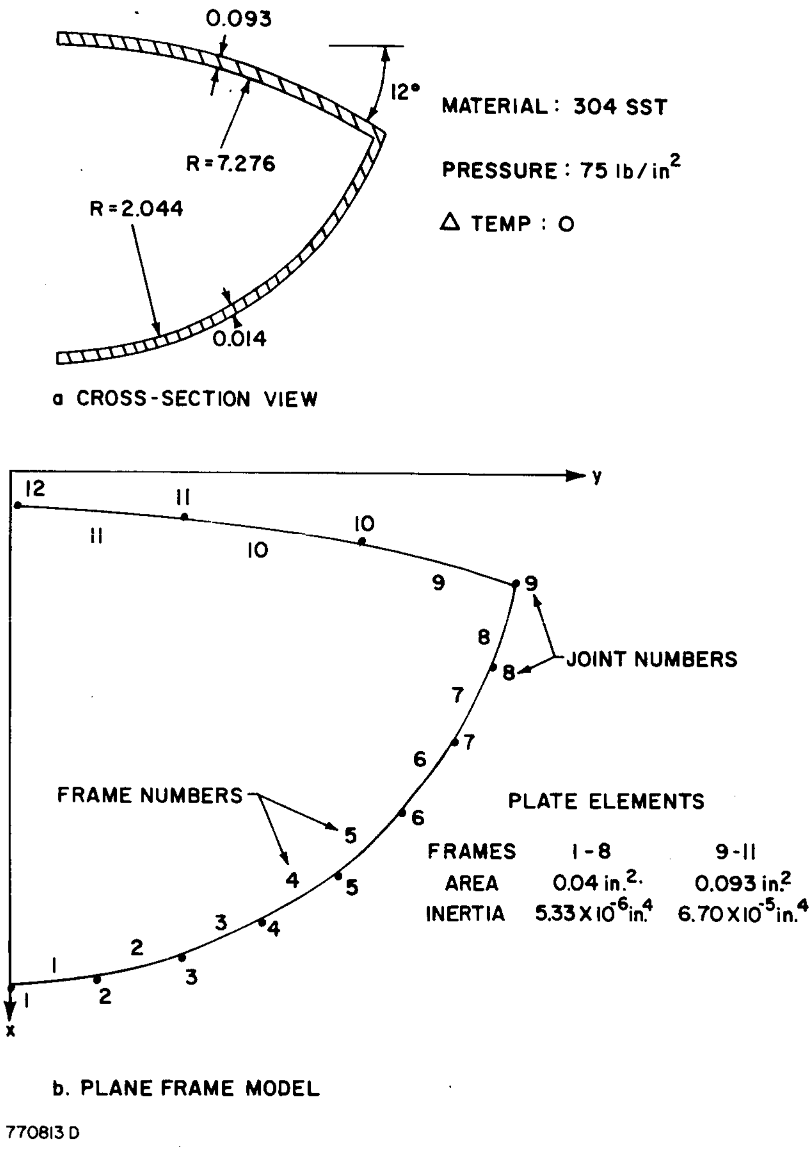


Figure C-3. CROSS SECTION AND ANALYTICAL MODEL OF PLENUM CHAMBER

TABLE C-I

RESULTS FROM "STRESS" PROGRAM PLENUM CHAMBER AT 75 LB/IN^2

Frame	Joint	Axial Load	Bending Moment	Primary Stress (lb/in^2)	Bending Stress (lb/in^2)	Secondary Stress (lb/in^2)
1	1	130.905	7.704	3272.625	28890.0	32.162
1	2	130.905	7.075	3272.625	26531.25	29.803
2	2	131.527	7.075	3288.175	26531.25	29.819
2	3	131.527	5.173	3288.175	19398.75	22.687
3	3	132.760	5.173	3319.000	19398.75	22.718
3	4	132.760	2.058	3319.000	7717. 50	11.036
4	4	134.569	2.058	3364.225	7717. 50	11.082
4	5	134.569	2.194	3364.225	8227. 50	11.592
5	5	136.915	2.194	3422.875	8227. 50	11.650
5	6	136.915	7.461	3422.875	27978.75	31.402
6	6	139.687	7.461	3492.175	27978.75	31.471
6	7	139.687	13.578	3492.175	50917.50	54.510
7	7	142.910	13.578	3572.750	50917.50	54.490
7	8	142.910	20.378	3576.750	76417.50	79.990
8	8	146.399	20.378	3659.975	76417.50	80.077
8	9	146.399	27.655	3659.975	19183.11	22.843
9	9	25.872	27.655	278.194	19183.11	19.461
9	10	25.872	55.276	278.194	38341.73	38.620
10	10	15.289	55.276	164.398	38341.73	38.501
10	11	15.289	105.349	164.398	73074.45	73.239
11	11	11.052	105.349	118.849	73074.45	73.193
11	12	11.053	122.085	118.849	84683.24	84.802

C. DETERMINATION OF STORAGE TANK AND PLENUM CHAMBER DYNAMICS

1. System Configuration and Vibration Loads

The configuration of the storage tank and plenum chamber used in the analysis is shown in Figure C-4. For this study, the only support was at the mounting ring, where support was considered continuous around the tank circumference.

The vibration requirements are shown in Table C-II.

TABLE C-II

VIBRATION REQUIREMENTS

1. Random Vibration	4 minutes $0.07g^2$/cps, uniform in frequency range 20–2000 cps	
2. Sinusoidal Vibration	(Logarithmic Sweep)	
Frequency (cps)	Duration (minutes)	Accelerations (g) (rms)
5–50	1.67	1.5
50–250	1.00	15.0
250–500	0.67	15.0
500–2000	1.0	7.5
100–130	0.25	45.0
550–650	0.50	36.0

2. Analysis: General

The two cases considered critical were evaluated: Case I, vibration in the longitudinal direction, Figure C-4: Case II, vibration in the lateral direction. For the analysis it was assumed that no liquid sloshing would occur.

Simple 1- or 2-degree-of-freedom models were formulated for both cases.

For the sinusoidal vibration input, the output acceleration $\ddot{y}_s$ is related to the input acceleration $\ddot{x}$ at resonance by the relation

$$\ddot{y}_s = \frac{\ddot{x}}{2\rho} ,$$

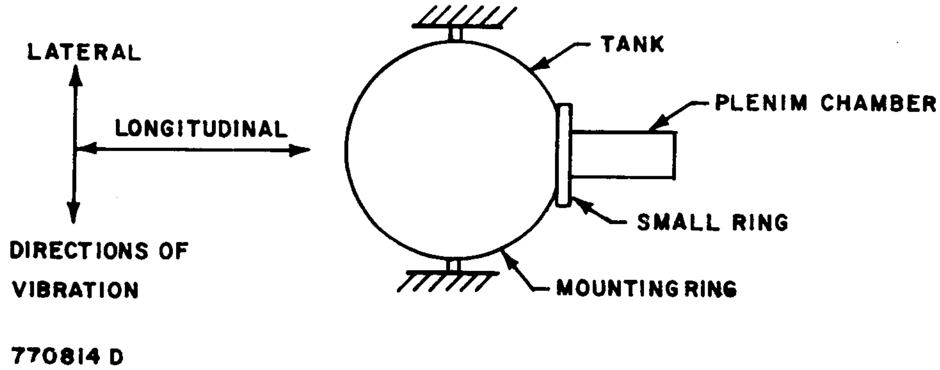


Figure C-4. STORAGE TANK AND PLENUM CHAMBER CONFIGURATION AND LOAD ORIENTATION where ρ is the damping factor.

For this study, a damping factor of from 0.01 to 0.05 was used. For the tank breathing mode, where the fluid is expected to increase damping, a factor of 0.05 was used. (Note: As the actual damping was unknown, its magnitude could be established by testing, and a more accurate analysis would be obtained.)

For the random vibration input, the output acceleration $\ddot{y}_r$ is related to the input power spectral density ($\mathcal{S}_f$) by the relation

$$\ddot{y}_r = 3 \sqrt{\frac{\pi}{2} \frac{f_n \mathcal{S}_f}{2\rho}} ,$$

where

f_n is the resonance frequency (cps), and
ρ is damping factor.

This gives the peak acceleration, which will not be exceeded 99.7 percent of the time.

3. Analysis: Case I

The following were types of motion included for Case I:

a) Tank breathing mode (filled with liquid)

b) Plenum-chamber—small-ring motion relative to support.

c) Motion of tank relative to the support (cantilevered on mounting ring).

The tank breathing mode was found by using as a reference, "Extensional Vibration of Axisymmetrical Shells," Hwang, AIAA Journal, January 1965, p. 23.

$$\omega^2 = \Omega^2 \frac{E}{\rho_a{}^2}$$

Ω = 0.870

E = 16×10^6 lb/in^3 (modulus of elasticity)

ρ = 0.160 lb/in^3 (density)

a = 8.76 inches (radius)

This results in $f \approx 3100$ cps.

To account for the liquid in the tank, the frequency is conservatively lower by the ratio

$$\sqrt{} = \sqrt{\frac{6.4}{57}} .$$

$\therefore f$ breathing mode (full tank) ≈ 1000 cps

$$\ddot{y}_r = 3 \sqrt{\frac{\pi}{2} \frac{f_n \; S_f}{2\rho}} = 3 \sqrt{\frac{\pi}{2} \frac{1000\,(0.00)}{2\,(0.05)}} = 100\ g$$

$$\ddot{y}_s = \frac{\ddot{x}}{2\rho} = \frac{7.5}{2\,(0.05)} = 75\ g .$$

The frequency associated with the motion of the plenum chamber and small ring relative to the support was found using influence coefficient data found in "Nonsymmetric Deformation of Dome-Shaped Shells of Revolution," by C. R. Steele, Journal of Applied Mechanics, June 1962.

From this information, the natural frequency f_n is 500 cps:

$$\ddot{y}_r = 3 \sqrt{\frac{\pi}{2} \frac{f_n \; S_f}{2\rho}} = 3 \sqrt{\frac{\pi}{2} \frac{(500)\,(0.07)}{2\,(0.01)}} = 15.8\ g$$

$$\ddot{y}_s = \frac{\ddot{x}}{2\rho} = \frac{15}{2\,(0.01)} = 750\ g.$$

Load = g (Wt); 750 (9. 1 lb) = 6825 pounds.

The frequency of the tank cantilever on the mounting ring was found by determining the deflection of the mounting ring (taken as a cantilevered beam), with the tank acting as a weight on the tip of the beam.

The corresponding frequency f_n = 1550 cps:

$$\ddot{y}_r = 3 \sqrt{\frac{\pi}{2} \frac{f_n \; S_f}{2\,\rho}} = 3 \sqrt{\frac{\pi}{2} \frac{(0.550)\,(607)}{2\,(0.05)}} \cong 120\ g$$

$$\ddot{y}_s = \frac{\ddot{x}}{2\rho} = \frac{7.5}{2\,(0.05)} = 75\ g .$$

4. Analysis: Case II

For vibration in the lateral direction, it was assumed that the frequency associated with the tank itself was above the 2000 cps range. Thus, the loadings associated with the tank and the fluid are essentially rigid body:

$$\ddot{y}_s = 45\ g .$$

The frequency associated with the plenum chamber and small ring was calculated by assuming the system as a mass attached to a portion of a hemisphere. Influence coefficient was calculated using results of "Non-symmetric Deformation of Dome-Shaped Shells of Revolution," by C. R. Steele, Journal of Applied Mechanics, June 1962. Resulting frequencies are of the order of 500 cps (rotation of this system) and 1100 cps (lateral translation of the system), which give critical loads:

$$\ddot{y}_r = 3\sqrt{\frac{\pi}{2}\,\frac{f_n\,\delta_f}{2\rho}} = 3\sqrt{\frac{\pi}{2}\,\frac{(1100)(0.07)}{2\,(0.01)}} \cong 235\ g$$

$$\ddot{y}_s = \frac{\ddot{x}_s}{2\rho} = \frac{7.5}{2\,(0.01)} = 375\ g\ .$$

With plenum chamber weight of 9.1 pounds (including small ring, plenum chamber, and components on the plenum chamber), loading is 9.1 (375), or 3400 pounds.

5. Summary of Critical Dynamic Loads

For Case I, vibration in the longitudinal direction, the plenum chamber imparts force of 6825 pounds at the small ring. Propellant loading is equivalent to 120 g.

In Case II, vibration in the lateral direction, the plenum chamber imparts force of 3400 pounds at plenum chamber c.g.; propellant loading is equivalent to 45 g.

D. DYNAMIC STRESS ANALYSIS

It is assumed that the maximum temperature does not exceed 80°F during the vibration of the system. This corresponds to an ammonia pressure of 153 psia. Since the structure is still in the atmosphere, a pressure differential of 140 lb/in^2 is assumed. The maximum amplification found in the dynamic analysis above is 120 g during longitudinal vibration. The dynamic analysis has been based on the model shown in Figure C-5.

The dynamic stress has been approximated by superimposing a gravity gradient, in the the ammonia, on a 140 lb/in^2 static analysis. The tank contains 57 pounds of ammonia, which is a mean density of slightly less than 0.02 $lb/in.^3$ Therefore, adding to the 140 $lb/in.^2$

$$(0.02\ lb/in^3)\ (120\ g)\ (\text{depth from surface}) = (0.02)\ (120)\ (17.101) = 41.04\ lb/in^2$$

at the pole point of Region 1.

This static condition was run on Avco Program 1888, Case 3.0, Memo A-3246,

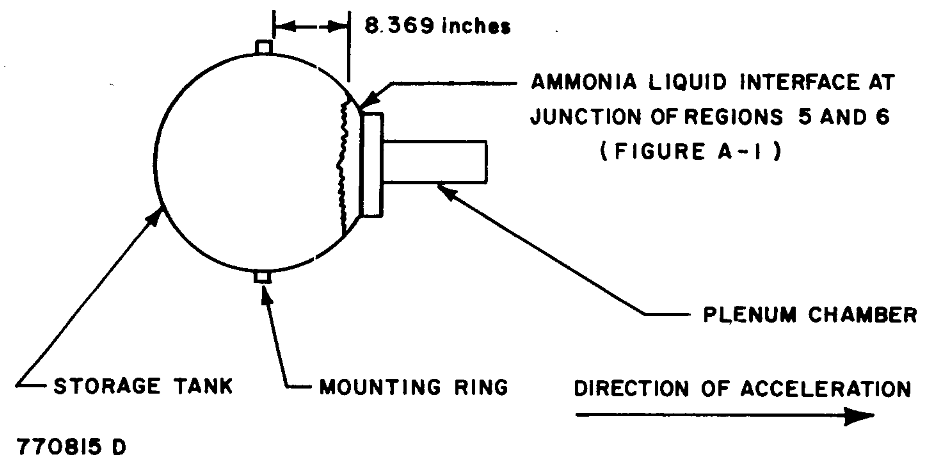


Figure C-5. DYNAMIC STRESS ANALYSIS MODEL

June 3, 1966. The H card reads as follows:

H TITANIUM AMMONIA TANK-140 psi/120 g/NO TEMP.

The 6825 pounds induced by the plenum chamber during longitudinal resonance was also included, even though this force does not occur at the same time that the ammonia mass is resonating. The inertia of the shell is also included.

The stress intensities computed with the above assumptions are plotted in Figure C-6. They are everwhere less than at design conditions (Figure C-2), thereby obviating analysis of the lateral vibration condition, which has lower dynamic loads.

Unlike the design condition static case, however, there is a reaction at the bolt circle flange, which was assumed fully restrained in this analysis; i.e., no deflection or rotation. This reaction leads to the following:

At the bolt circle

Moment = 103.4 inches lb/in

Bending stress = 39.7 klb/in^2

Total bolt load = 14,900 pounds

At the base of the flange

Moment = 67.9 inches lb/in

Bending stress = 26.1 klb/in^2

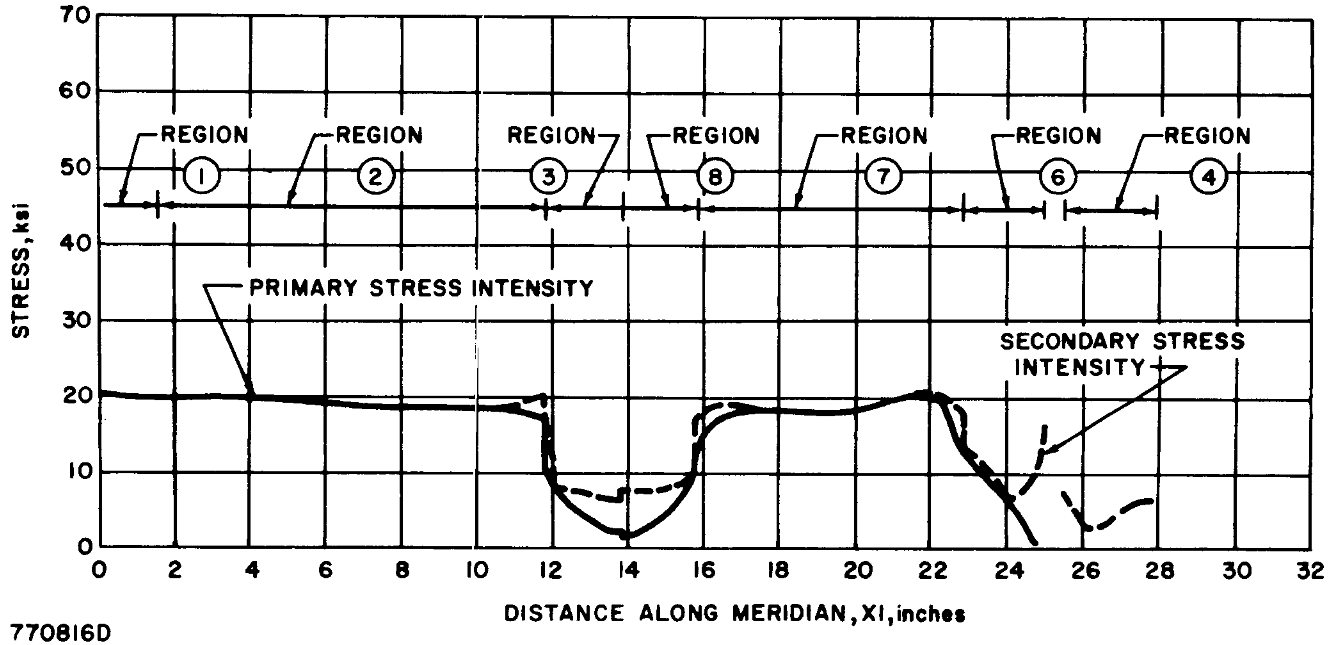


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Figure C-6. AMMONIA STORAGE TANK STRESS INTENSITY AT PEAK AMPLIFICATION OF LONGITUDINAL EXCITATION PLUS 140 LB/IN^2

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